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STRESS STATE IN THE COMBINED

STRESS TORSION TEST

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by

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Fracture of Metals During Deformation Processing
Under Conditions of Hot Working
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ABSTRACT

An analysis of the stresses in the axial stress-torsion test is presented by considering the gage section as a cylinder undergoing uniform axial deformation with superimposed torsion. Calculation of the stresses requires measurement of the radial deformation and penetration of the plastic region into the notch shoulders.

The hydrostatic stress exhibits a peak within the wall of the gage section which increases with decreasing gage length. The fracture strain in room temperature tests increases with decreasing gage length when a compressive axial load is applied and decreases slightly with decreasing gage length when a tensile axial load is applied. Severe distortion and large changes in length of the gage section during hot torsion testing prevent application of the analysis to high temperature tests.

Test data are presented for hot-rolled Inconel 600 at room temperature and 1800F.

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INTRODUCTION

The torsion test has a number of advantages for simulating deformation processes. The chief advantage is that large plastic strains, comparable to those found in metalworking processes, can be achieved without complications to the stress state due to necking. Moreover, tests at constant and high deformation rate are made readily by twisting at a constant rotational velocity. The chief disadvantage of the torsion test is that stress, strain, and strain rate vary from the axis to the outer fiber of a solid cylindrical specimen. However, this problem can be largely eliminated by using a hollow cylindrical specimen with a relatively thin wall thickness.

The state of stress in the simple torsion test consists of pure shear, with equal tensile and compressive stresses at 45° to the shear stresses. A variation in the stress state can be achieved if axial forces, tensile or compressive, are superimposed on the twisting moment. This type of combined stress torsion test would be particularly useful in establishing the stress criteria for fracture in metal deformation. Bridgman(1) first performed a combined stress torsion test at room temperature with steel and showed large increases in shear strain to fracture with high compressive stresses superimposed on the shear stress. More recently the combined stress torsion test has been used to study ductile fracture during room temperature deformation(2)(3).

The combined stress torsion test was first applied to fracture under hot working conditions during previous research under this project(4). Figure 1 shows the effect of stress state on shear strain to fracture for Inconel 600 tested at various temperatures. The stresses plotted as σ_{max}/τ_{max} represent the largest values of normal and shearing stress applied during the test, assuming that they act on a plane normal to the cylinder axis. Usually this is the initial axial stress and the shearing stress corresponding to the peak in the torque-twist curve. Because the complete analysis of stress in a plastic thin-wall cylinder of short gage length with elastic constraints at the shoulders is not known for combined torque and axial load it was not possible to calculate the maximum principal stress and the maximum shear stress. Nevertheless, the data in Figure 1 show a strong influence of compressive stress in increasing the ductility during hot working.

More recently the combined stress torsion test has been used to study in detail the fracture mechanisms in warm work and hot work in pure nickel(5). It has also been used effectively to study the hot workability of the complex superalloys Waspaloy and Inconel 718 and to correlate their workability with extrusion conditions(6).

STRESS STATE AND DEFORMATION IN COMBINED STRESS TORSION TEST

Since the tensile component of hydrostatic stress promotes the initiation of fracture at sites of structural irregularity while a compressive hydrostatic component suppresses crack growth, it is apparent that a better understanding of the stress distribution in the gage length of the combined stress torsion test would greatly increase its usefulness.

In most previous work utilizing this test(1)(4)(6) only the axial and torsional shear stresses were considered. These were taken to be uniformly distributed over the cross-section and are given by

$$\sigma_Z = \frac{F}{2\pi\bar{r}t} \quad \text{(Axial Stress)}$$
$$\tau_{Z\theta} = \frac{T}{2\pi\bar{r}^2 t} \quad \text{(Torsional Stress)} \qquad [1]$$

where F = axial force (lbs.)
T = torque (in-lbs.)
$\bar{r}$ = mean radius (in.)
t = wall thickness (in.)

See Figure 2 for the typical geometry of the test specimen gage length

A notable exception to this is an analysis(3) which concludes that, in addition to the stresses in Equation [1], a circumferential stress exists, given by

$$\sigma_\theta = \sigma_Z - \frac{2\tau_{Z\theta}}{\bar{r}\Delta\theta/\Delta\ell} \qquad [2]$$

where $\Delta\theta/\Delta\ell$ is the increment of twist per change in gage length. The above analysis is based on the observation that the mean radius of the gage section of the specimens tested does not change during the test. This behavior, however, is not typical of all torsion specimens. Furthermore, the distribution of stresses is not found.

In earlier work by another investigator(7), the radial distribution of stresses in a solid bar under torsion and tension were found using the principle of minimum work. These results, however, cannot be extended to hollow torsion specimens.

Ideally, to establish the radial distribution of stresses and the transiton from plastic to elastic deformation at the notch shoulders, an exact elastic-plastic analysis is required. In the following, a less sophisticated approach to the problem is presented which is based on an approximation of the deformation in the gage length. Nevertheless, the analysis is sufficiently accurate to provide an insight into the nature of the distribution of stresses and the influence of the gage section geometry on the stresses.

It has previously been reported(1), and observed in the present tests as well, that the radial expansion or contraction of the gage section under torsion and axial load is nearly uniform along the gage length. Therefore, as a first approximation, the gage section can be considered as a hollow cylinder undergoing uniform twist and extension or compression with accompanying radial contraction or spread. The shoulders at each end of the gage length act as rigid ends partially constraining radial deformation. This constraint develops radial shear stresses at each end of the gage length, as depicted in Figure 3.

Based on these assumptions and the incompressibility condition for plastic deformation, the radial deformation rate is given by

$$\dot{u} = -\dot{\varepsilon}_z(r^2 - r_n^2)/2r \qquad [3]$$

where $\dot{\varepsilon}$ = axial strain rate

r = radial coordinate

r_n = neutral radius (position with no deformation)

The concept of the neutral radius is introduced since simultaneous outward spread of the outer surface and inward spread of the inner surface is observed in some tests, indicating that there is no displacement at some internal point.

Two previous studies using hollow cylindrical specimens indicate the extent of radial constraint provided by the shoulders and serve as bounds on the type of stress states to be expected in axial stress-torsion tests. First, the long, thin-walled tubes used by Taylor and Quinney(8) in their studies of yielding under combined stresses are not subject to radial constraint and the deformation is completely in one direction with the neutral radius at the cylinder axis (r_n = o). As a result, the radial and circumferential stresses are zero and the axial and torsional stresses are given by Equation [1]. This stress state would not be expected in axial stress-torsion tests except for those with long, thin-wall gage sections.

Second, tubes with a circular notch tested under axial loads by Bridgman(9) essentially have no gage length but the gage section is subject to full constraint by the material outside the notch. In this case the diameter does not change during the test, and the neutral radius is the mean radius ($r_n = \bar{r}$). The stresses in the notch given below

$$\sigma_r = \sigma_{Za} \ln[1 + \frac{a}{2R}(1 - x^2/a^2)]$$

$$\sigma_\theta = \sigma_r + \sigma_{Za}/2 \qquad [4]$$

$$\sigma_Z = \sigma_r + \sigma_{Za}$$

where $2a$ = wall thickness, t (in.)

R = notch radius (in.)

x = distance measured from center of tube wall (in.)

σ_{Za} = axial stress at each edge (psi)

This stress state is characterized by a nearly-parabolic radial stress distribution, while the circumferential and axial stresses have the same distribution superimposed on uniform stresses $\sigma_{Za}/2$ and σ_{Za}, respectively. Stresses of this nature would be approached in specimens with very small gage lengths.

The stress state in an actual axial stress-torsion test can be expected to lie somewhere between the two extremes described above, depending on the gage length and wall thickness of the specimens.

ANALYSIS

Returning to the expressions for the assumed deformation, Equation [3], the radial and circumferential strain rates become

$$\dot{\varepsilon}_r = d\dot{u}/dr = -\dot{\varepsilon}_Z(1 + r_n{}^2/r^2)/2$$

$$\dot{\varepsilon}_\theta = \dot{u}/r = -\dot{\varepsilon}_Z(1 - r_n{}^2/r^2)/2 \qquad [5]$$

Substituting these expressions in the Levy-Mises equations, and rearranging, the stress differences are found:

$$\sigma_\theta - \sigma_r = \frac{2}{3}\frac{\dot{\varepsilon}_Z}{\lambda}\frac{1}{r'^2}$$

$$\sigma_r - \sigma_Z = -\frac{\dot{\varepsilon}_Z}{\lambda}(1 + \frac{1}{3r'^2}) \qquad [6]$$

$$\sigma_Z - \sigma_\theta = \frac{\dot{\varepsilon}_Z}{\lambda}(1 - \frac{1}{3r'^2})$$

where $r' = r/r_n$

and $\lambda = \dot{\bar{\varepsilon}}/\bar{\sigma}$, ratio of effective strain rate to effective stress at the current value of total effective strain

For plastic flow, a yield criterion must also be satisfied. The von Mises yield criterion will be used

$$(\sigma_\theta - \sigma_r)^2 + (\sigma_r - \sigma_Z)^2 + (\sigma_Z - \sigma_\theta)^2 + 6\tau_{Zr}{}^2 + 6\tau_{Z\theta}{}^2 = 2Y^2$$

where Y is the flow stress of the material in simple tension. Substituting the expressions for stress differences into this yield criterion

$$(1 + \frac{1}{3r'^4})^{1/2} \frac{\dot{\varepsilon}_Z}{\lambda} = (Y^2 - 3\tau_{Z\theta}{}^2 - 3\tau_{Zr}{}^2)^{1/2} \qquad [7]$$

Considering the right side of Equation [7] as a reduced effective yield stress, σ_e, the equations for stress differences can now be written as

$$\begin{aligned} \sigma_\theta - \sigma_r &= \frac{2}{\sqrt{3}} \sigma_e/(3r'^4 + 1)^{1/2} \\ \sigma_r - \sigma_Z &= -\frac{\sigma_e}{\sqrt{3}} (3r'^2 + 1)/(3r'^4 + 1)^{1/2} \\ \sigma_Z - \sigma_\theta &= \frac{\sigma_e}{\sqrt{3}} (3r'^2 - 1)/(3r'^4 + 1)^{1/2} \end{aligned} \qquad [8]$$

To establish the expressions for the individual stresses the first of Equation [8] is substituted into the equation for radial equilibrium

$$\sigma_\theta - \sigma_r = r(\frac{\partial \sigma_r}{\partial r} + \frac{\partial \tau_{Zr}}{\partial Z}) = \frac{2}{\sqrt{3}} \sigma_e/(3r'^4 + 1)^{1/2} \qquad [9]$$

Before this equation can be integrated to find σ_r, the variation of σ_e and τ_{Zr} must be specified. σ_e varies with total effective strain and the torsional shear stress, both of which will be constant for large strains. Since large strains are involved in the tests, σ_e will be considered constant. τ_{Zr} is the radial shear stress which develops because of the constraint supplied by the shoulders at each end of the gage section. The distribution and magnitude of the radial shear stress is unknown, but, since it is of equal magnitude and opposite sign at each end of the gage section, the axial gradient will be taken as constant

$$\frac{\partial \tau_{Zr}}{\partial Z} = 2\ \tau/\ell \qquad [10]$$

where τ = radial shear stress at each end

ℓ = gage length

The distribution of the radial shear stress at each end must be such that the direction of the shear stress opposes the direction of the radial spread. Two different distributions are considered:

(a) Constant radial shear stress

$$\tau = \begin{cases} \pm k \text{ for } r_i<r<r_n \\ \mp k \text{ for } r_n<r<r_o \end{cases} \qquad [11a]$$

where r_i, r_o = inside, outside radii

Upper sign for axial compression.

Lower sign for axial tension.

(b) Since the radial shear stress is proportional to $\frac{\partial u}{\partial Z}$ which is zero at the neutral radius and has maximum values at the inner and outer surfaces, the radial shear stress is probably more accurately represented by a linear function

$$\tau = \mp k \left\{\frac{r - r_n}{r_n}\right\} \qquad [11b]$$

Upper sign for axial compression.

Lower sign for axial tension.

These shear stress distributions are shown in Figure 3.

Using Equations [10] and [11] and assuming σ_e is constant, Equation [9] can be integrated to give

$$\sigma_r = \frac{\sigma_e}{\sqrt{3}} \ln(R_i/R) \mp \frac{2kr_n}{\ell} \begin{cases} (a)\begin{cases} r' - r_i' \text{ for } r_i<r<r_n \\ 2 - r' - r_i' \text{ for } r_n<r<r_o \end{cases} \\ (b)\ r' - r_i' - \frac{1}{2}(r'^2 - r_i'^2) \end{cases} \qquad [12]$$

where $R = (1 + (1 + 3r'^4)^{1/2})/r'^2$

At this point the radial stress can be calculated if the magnitude of the radial shear stress and the neutral radius, k and r_n, are known. Neither quantity is known, but a relationship between them can be obtained by applying the boundary condition $\sigma_r = 0$ at the outside radius

$$\frac{\sigma_e}{\sqrt{3}} \ln(R_i/R_o) = \pm \frac{2kr_n}{\ell} \begin{cases} (a)\begin{cases} r_o' - r_i' \text{ if } r_n<r_i \\ 2 - r_o' - r_i' \text{ if } r_i<r_n<r_o \end{cases} \\ (b)\ r_o' - r_i' - \frac{1}{2}(r_o'^2 - r_i'^2) \end{cases} \qquad [13]$$

The variation of the magnitude of the radial shear stress k with neutral radius is shown in Figure 4. It is evident that when the neutral radius is zero the radial shear stress is zero (free radial deformation). As the neutral radius approaches the mean radius, the shear stress becomes unbounded (full constraint of radial deformation). Furthermore, for a given value of shear stress, as the gage length ℓ decreases, the neutral radius approaches the mean radius, while for increasing ℓ the neutral radius approaches zero. This result is consistent with the free radial deformation observed in the long, thin tubes used by Taylor and Quinney(8) and the fully-contrained deformation in Bridgman's notched tubes under axial load(9).

To eliminate the problem presented by this unknown shear stress magnitude, the factor $2kr_n/\ell$ is eliminated in Equation [12] by using Equation [13], giving

$$\frac{\sigma_r}{\sigma_e/\sqrt{3}} = \ell n(R_i/R) - \ell n(R_i/R_o) \begin{cases} \text{(a)} \begin{cases} (r' - r_i')/(r_o' - r_i') \text{ for all } r \text{ if } r_n<r_i \\ \left.\begin{array}{l} (r' - r_i')/(2 - r_o' - r_i') \text{ for } r<r_n \\ (2 - r' - r_i')/(2 - r_o' - r_i') \text{ for } r>r_n \end{array}\right\} \text{if } r_n>r_i \end{cases} \\ \text{(b)} \quad r' - r_i' - 1/2(r'^2 - r_i'^2)/r_o' - r_i' - 1/2(r_o'^2 - r_i'^2) \end{cases} \qquad [14]$$

Thus, if r_n is known, the radial stress σ_r can be calculated.

Under the assumptions described previously the neutral radius can be calculated from Equation [3] if measurements of the deformation of the gage section are made.

$$r_n = r_o\left(1 - 2\,\frac{\ell}{\Delta\ell}\,\frac{\Delta d_o}{d_o}\right)^{1/2} \qquad [15]$$

where $\Delta\ell$, Δd_o are change in gage length and outside diameter, respectively.

The following procedure can now be used to calculate the stresses in the axial stress-torsion test:

(i) From measurements of the change in gage length and change in outside diameter during the test, calculate the neutral radius from Equation [15].

(ii) Calculate the radial stress distribution from Equation [14], using both assumed radial shear stress distributions.

(iii) Calculate the axial and circumferential stress distributions from Equation [8].

This procedure was used to calculate the stress distributions in torsion specimens of various gage lengths for deformation at room temperature.

EXPERIMENTAL RESULTS AND CALCULATIONS

Test were conducted at room temperature on torsion specimens of Inconel 600 with gage lengths of 1/4-, 1/8-, 1/16- and 1/32-inch. The material was in the as-received hot rolled condition with a grain size of approximately .01 mm. For each gage length, three specimens were tested in torsion, one each at no axial load, high compressive axial load, and high tensile axial load. The axial loads were applied such that the average axial stress was one-half the yield stress of the material.

The tests were conducted at low strain-rate ($\sim .01 sec^{-1}$) so that the outside diameter and gage length could be measured as the test progressed. The change of outside diameter with twist is shown in Figure 5 for the tests with axial load. Also shown in the figure is the change in diameter which would occur in the two extreme cases: $r_n = 0$ and $r_n = \bar{r}$. The deformation for the 1/4- inch specimens is nearly that for free radial deformation ($r_n = 0$), but the deformation of the 1/32-inch specimens is less than that for $r_n = \bar{r}$ (full constraint of radial deformation). This indicates that the plastically-deforming material which contributes to the radial deformation exceeds the amount in the gage section and the region of plastic deformation extends into the notch shoulders.

Therefore, the deformed specimens were cut longitudinally and microhardness measurements were made to determine the extent of plastic deformation into the notch shoulders. The transition between plastic and non-plastic material is gradual, but the boundary of the plastic deformation region for specimens of all gage lengths is shown in Figure 6. Plastic deformation in the 1/4- inch specimens extends into the shoulders very little, but the penetration increases substantially as the gage length decreases. In fact, the actual gage length (length of plastic deformation) in the 1/32- inch specimens is more than three times the geometric gage length. Now when the actual gage length is used in Equation [15], the line corresponding to $r_n = \bar{r}$ in Figure 5 is less than that measured for the 1/32- inch specimens. It should be pointed out that even though the plastic deformation boundary is curved, the ensuing radial variation in axial deformation is small compared to the overall axial deformation so that the original assumption of uniform axial strain holds.

With the actual values of the gage length determined in the microhardness tests and the measured values of radial deformation, the procedure outlined above was used to calculate the stress distributions in the axial sttess-torsion specimens with geometric gage lengths of 1/4-, 1/16- and 1/32- inch. The results of these calculations are shown in Figure 7. For comparison, the stresses in the long, thin-walled tubes used by Taylor and Quinney(8) and the notched cylinders used by Bridgman(9) are included. It can be seen that the stresses exhibit a peak which increases in magnitude as the gage length decreases. The hydrostatic stress, plotted as the dotted line in Figure 7, also exhibits a peak which increases as the gage length decreases, even though the average value of the hydrostatic stress increases very little.

If hydrostatic pressure suppresses crack propagation, it would be expected from the above results that cracks which form at the outer surface of small gage length compression-torsion tests will not grow readily through the wall because of the increased hydrostatic pressure within the wall. On the other hand, cracks forming in large gage length specimens will not be retarded because the hydrostatic pressure is nearly constant through the wall. The converse would be expected in tension-torsion specimens. The high hydrostatic tension peak in small gage length specimens would promote crack formation more readily than the nearly uniform hydrostatic stress in large gage length specimens.

Since the point of maximum torque during a torsion test has been shown previously to coincide with the initiation of fracture(4), the shear strain at maximum torque can be used as a measure of strain to failure. For room temperature deformation this is close to the strain to complete fracture. To determine the effect of gage length, and thus hydrostatic stress, on crack growth during deformation, the shear strain at maximum torque is plotted as a function of gage length for the room temperature tests on Inconel 600 described above. The shear strain at maximum torque was calculated from

$$\gamma_{TM} = \theta r/\ell$$

θ = angular rotation at maximum torque (radians)
r = outside radius at maximum torque (in.)
ℓ = actual gage length as determined by hardness tests on deformed specimens (in.)

The results plotted in Figure 8 show a sharp increase in fracture strain in specimens of small gage length with axial compression. The decrease in fracture strain in specimens with axial tension is not nearly as large. Similar results have been observed in previous work on the combined stress torsion test(3)(4), i.e., for a given magnitude of axial stress the increase in fracture strain due to axial compression is much greater than the decrease in fracture strain due to axial tension (see Figure 1).

ELEVATED TEMPERATURE TESTS

An attempt was made to apply the results of the preceding analysis to axial stress-torsion tests at temperatures in the hot working range. In a program similar to the room temperature tests, specimens of various gage lengths were tested in pure torsion, torsion-tension and torsion-compression at 1800°F. To facilitate measurement of the radial deformation and change in gage length, the tests were performed by repeated cycles of heating to 1800°F, twisting through about 30° of rotation at a strain rate of $.01 sec^{-1}$. The specimens were then quenched in water to room temperature, and measurements were made of diameter and length of the test section. However, during the tests the gage sections became severely distorted and the deformation along the gage length was highly non-uniform. In addition, large deviations from the preset axial load occurred as twisting progressed. In the pure torsion tests, the gage section contracted a large amount, inducing a tensile axial force. To maintain the desired axial load, it was necessary to displace the tail stock during the test to compensate for the change in gage length.

Since this mode of deformation was not accounted for in the analysis described above, the analytical results cannot be applied to the high-temperature tests. Since 1800°F is in the hot working region for Inconel where extensive r-type cavity formation and link-up will occur(4)(5) and where high-temperature restoration processes such as recrystallization and grain growth will be active, it is not surprising that well behaved deformation modes were not observed. More extensive research over a wider range of specimen geometry, especially with larger diameter specimens, is required to determine the possibility of extending this stress analysis to the high temperature hot working region.

AXIAL STRAIN DURING TORSION

In pure torsion of cylindrical specimens, the material in the gage section is subjected to simple shear and no normal strains would be expected. However, in the room temperature tests described previously small axial elongation (~10%) were observed, even though a pure twisting torque was applied to the specimen. In the high temperature tests, large contractions of the gage section were observed during twisting.

The axial elongation found for the room temperature torsion tests is plotted in Figure 9 for each of the pure torsion specimens. The elongation varies linearly and the ratio of axial strain to shear strain is indicated for each curve. Similar results have been reported by other investigators(10)(11).

Axial elongations during twist in room temperature tests should be explainable on the basis of the anisotropy of the material. In the torsion test specimens, a preferred orientation is developed in the circumferential direction due to the progressing twist. This results in a differential in workhardening between the torsional and axial modes of deformation. According to an analysis by Hill(12), for large twist an axial elongation will develop if $\sqrt{3}$ x torsional shear stress is greater than the axial yield stress. For an isotropic material, these will be equal.

In the high-temperature (1800°F) torsion tests severe contraction of the gage section occurred during twisting, as shown in Figure 10. This behavior has been observed in other hot torsion tests in the hot working region (13)(14) using solid torsion specimens. In order to extend the analysis of the stress state in the hot torsion test to account for these effects it will be necessary to identify the mechanisms responsible for these length changes. It would seem possible to correlate the onset and magnitude of the length changes with the fracture and restoration processes occurring at different temperatures and strain rates.

SUMMARY AND CONCLUSIONS

Based on the assumption that the gage section in axial stress-torsion tests undergoes uniform twist as well as uniform axial strain, the stress distributions in the gage section are derived. Calculation of the stresses requires measurement of the radial deformation and extent of the plastic deformation into the notch shoulders. The spread of the plastic region into the notch shoulders increases with decreasing geometric gage length. The radial distribution of stresses exhibits a peak whose amplitude increases with decreasing gage length. In room temperature tests, the fracture strain varies in accordance with the calculated hydrostatic stress.

The analysis of the stresses in the axial stress-torsion test is based on an approximation of the deformation and an assumed distribution of the radial shear stress which partially constrains radial deformation of the material. For each of the two radial shear distributions assumed, the nature of the resulting normal stress distributions is the same. The analytical results thus give a good indication of the variation of the hydrostatic stress with gage length in the axial stress-torsion test. Non-uniform deformation along the gage length and marked changes in length of the specimen preclude the application of this analysis to hot torsion testing without more detailed knowledge of the processes involved.

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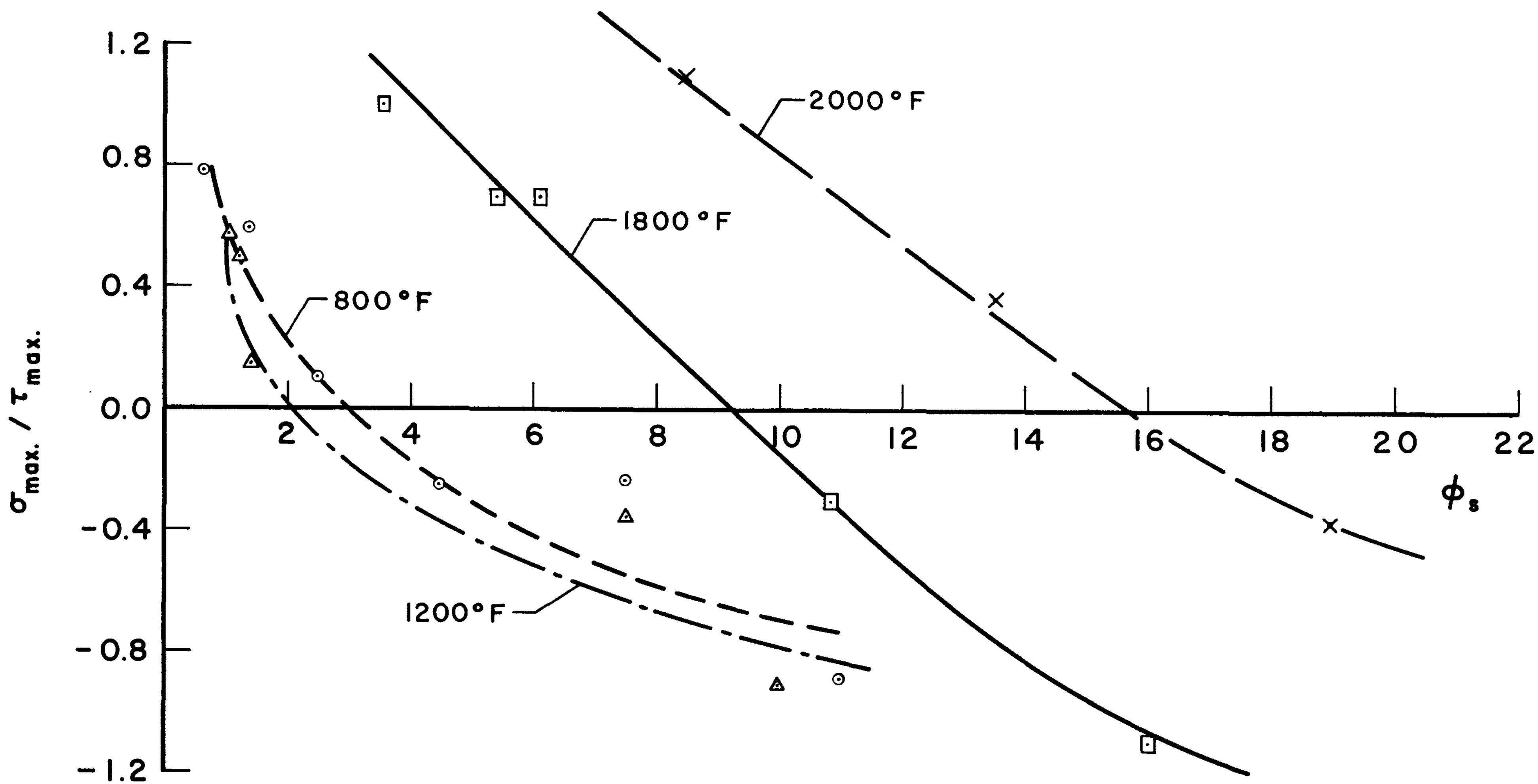


FIG 1. RATIO OF MAXIMUM NORMAL STRESS TO MAXIMUM SHEAR STRESS vs. ϕ_s FOR HOT-ROLLED INCONEL 600

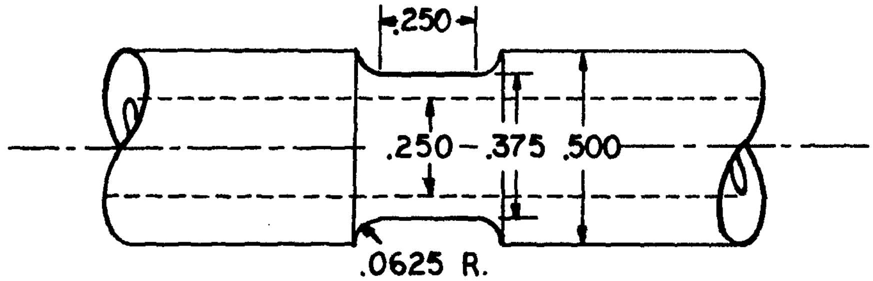


FIG 2. DIMENSIONS OF A TYPICAL GAGE SECTION

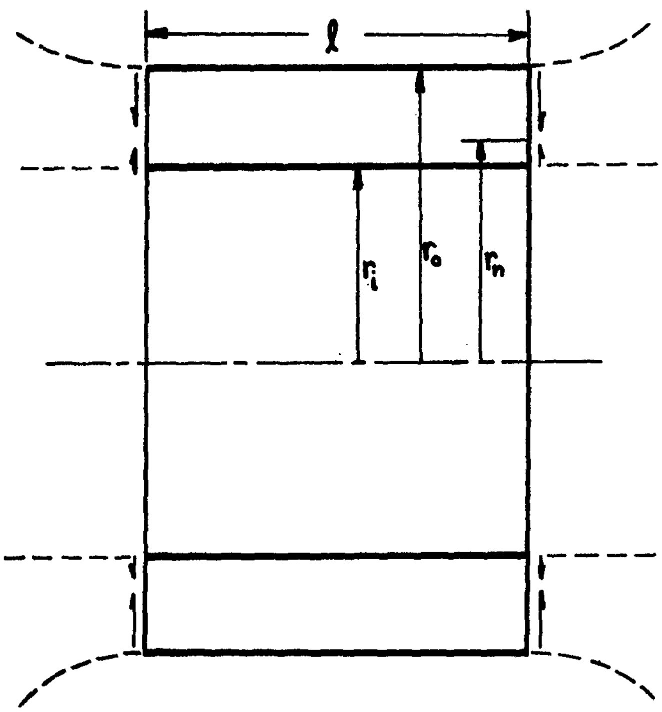
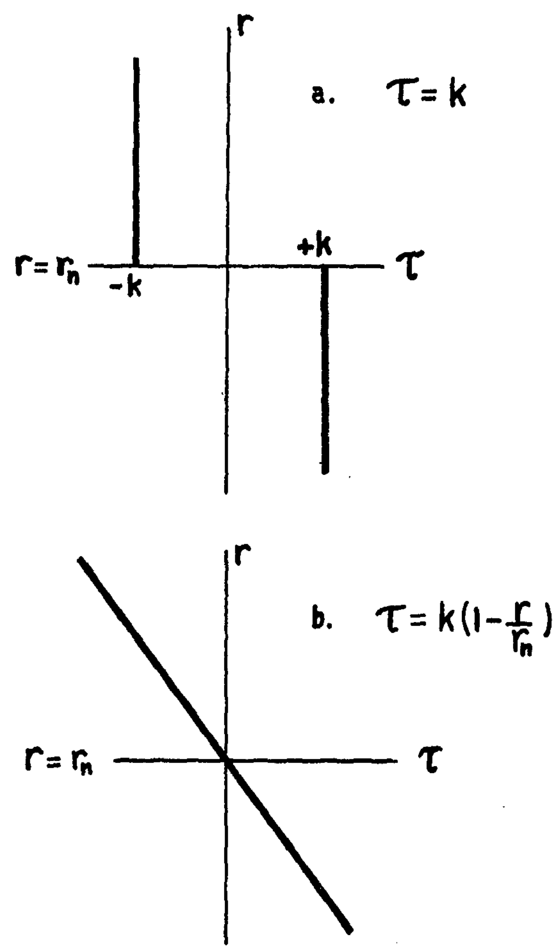


FIG 3. RADIAL SHEAR STRESS AT EACH END OF THE GAGE SECTION. SHEAR DIRECTION SHOWN FOR AXIAL COMPRESSION.



ASSUMED SHEAR DISTRIBUTIONS

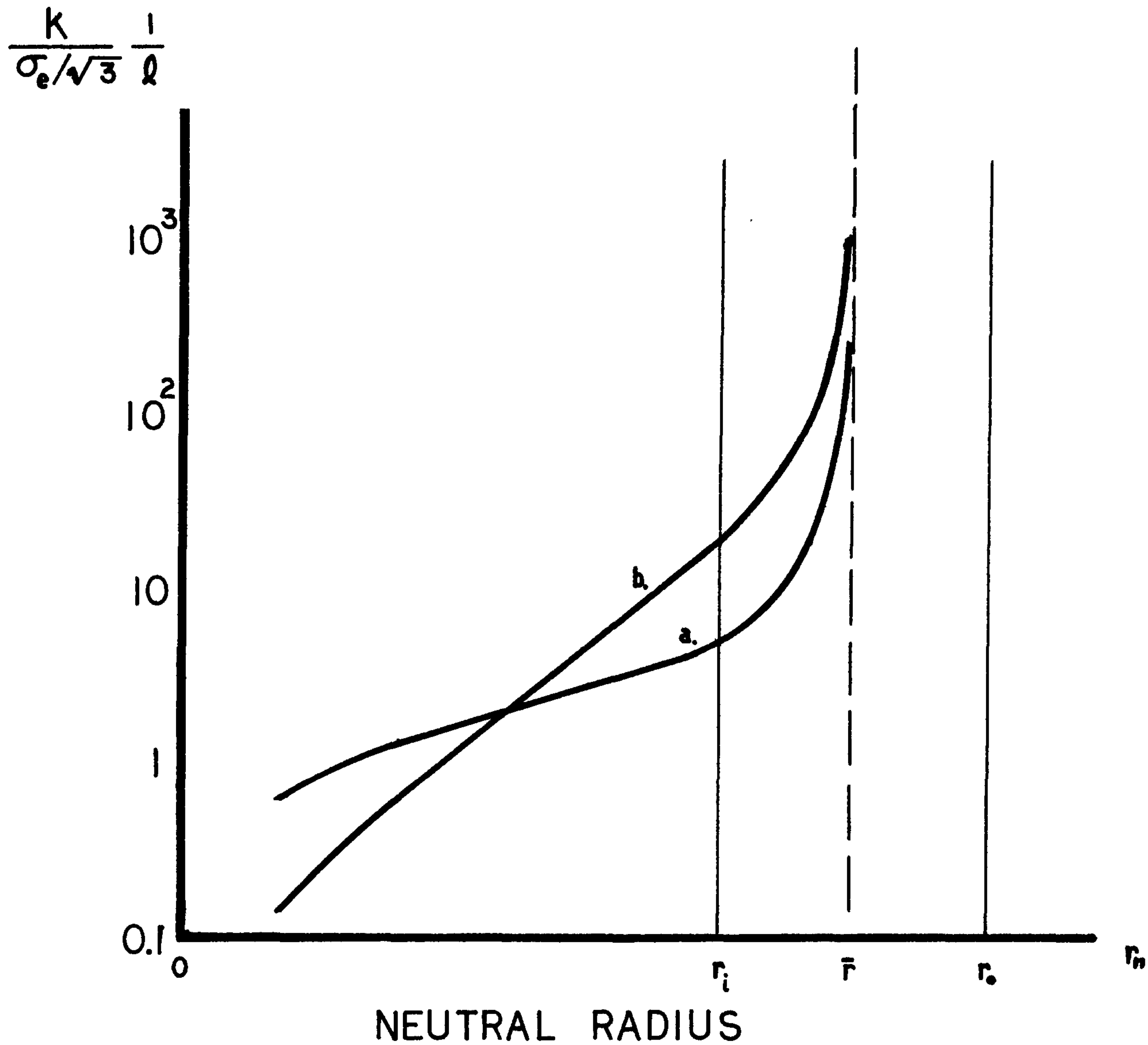


FIG 4. $\frac{K}{\sigma_e/\sqrt{3}}\frac{1}{\ell}$ vs. NEUTRAL RADIUS

FOR THE CASE r_i .125″ r_o .1875″ $\bar{r}$ .15625″

a. $\tau = K$

b. $\tau = K(1 - r/r_n)$

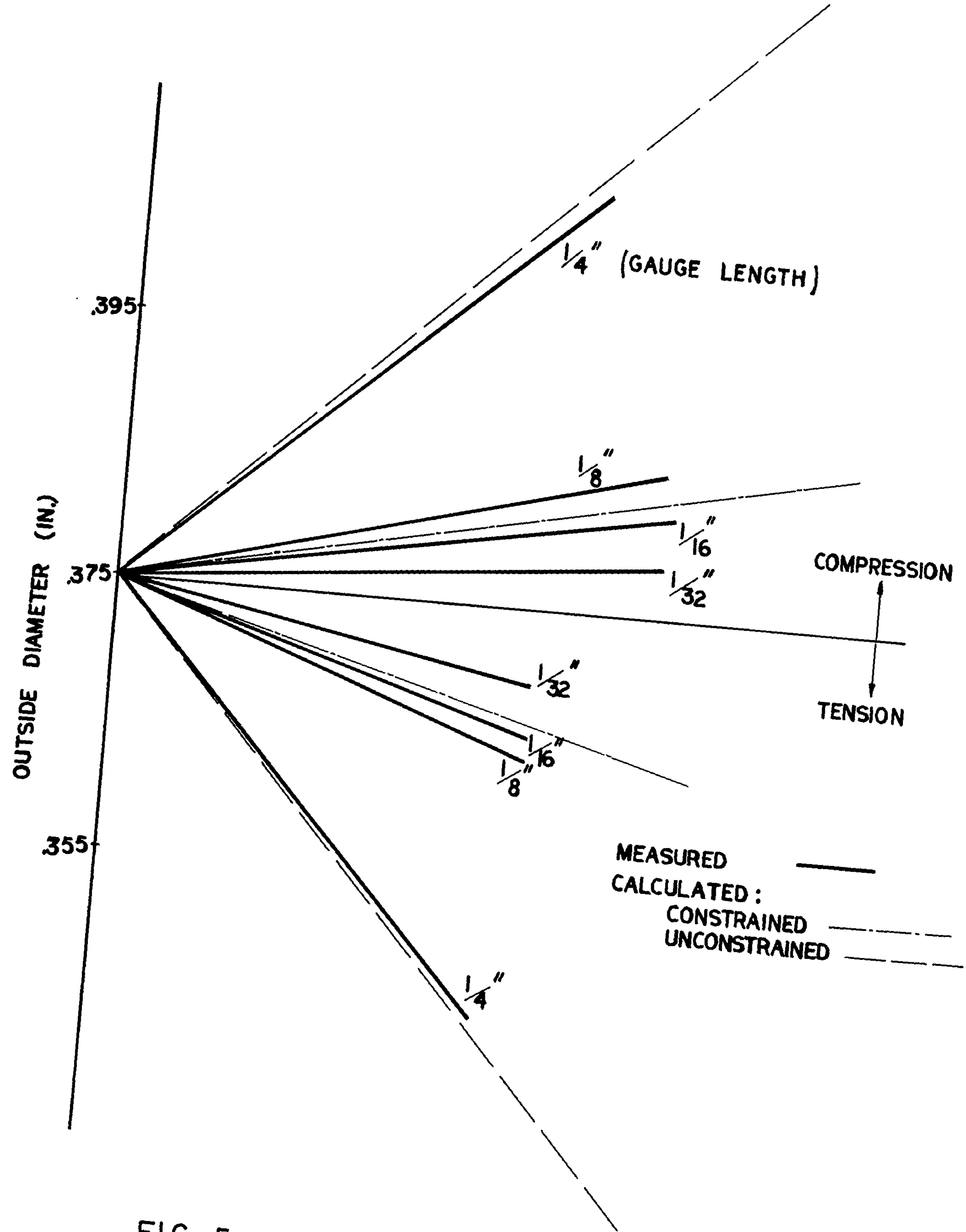


FIG 5. CHANGE IN OUTSIDE DIAMETER WITH ROTATION FOR INCONEL TESTED AT ROOM TEMPERATURE

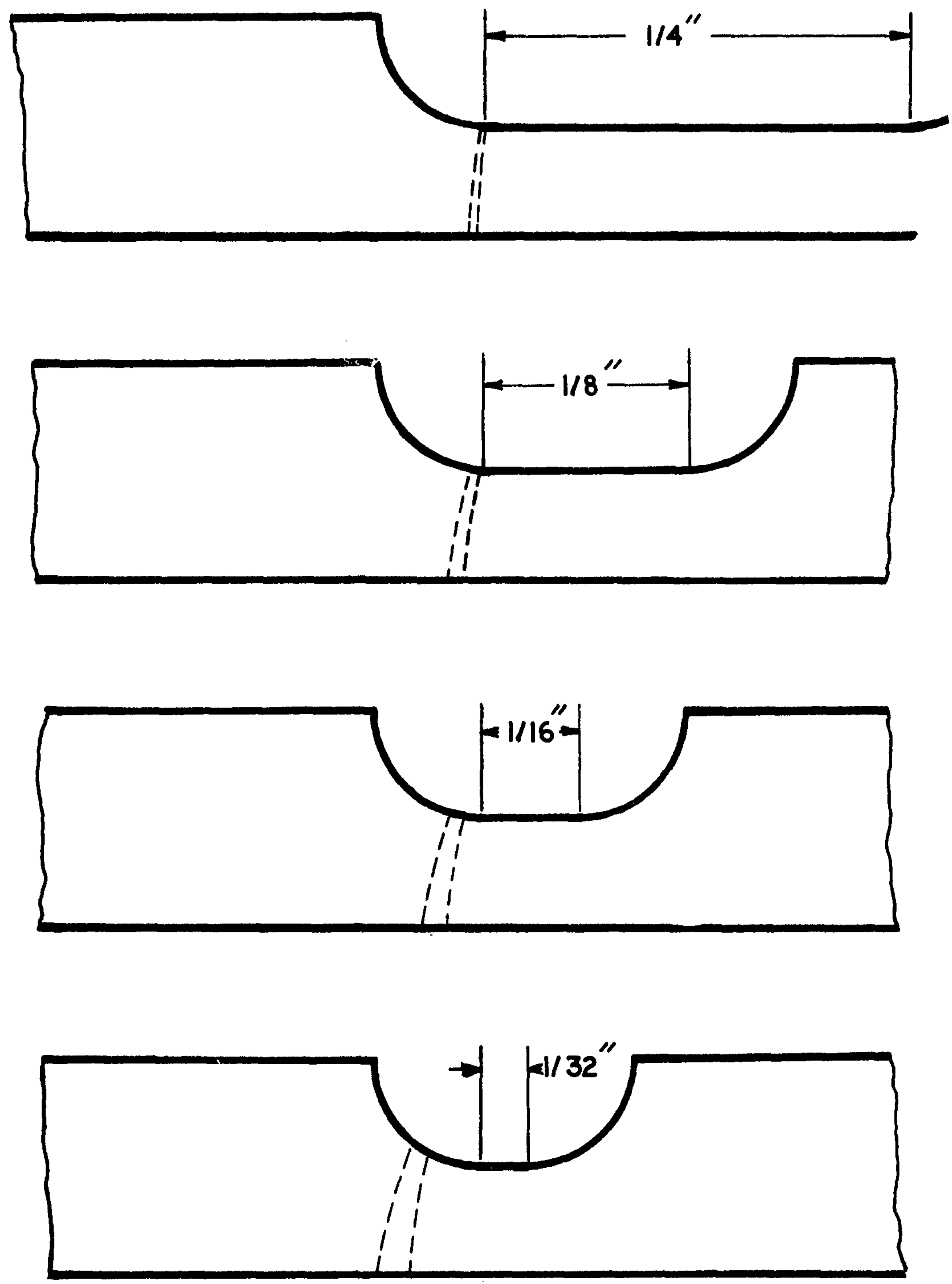


FIG 6. PENETRATION OF THE PLASTIC REGION INTO THE NOTCH SHOULDERS. DASHED LINES INDICATE TRANSITION FROM PLASTIC TO ELASTIC DEFORMATION.

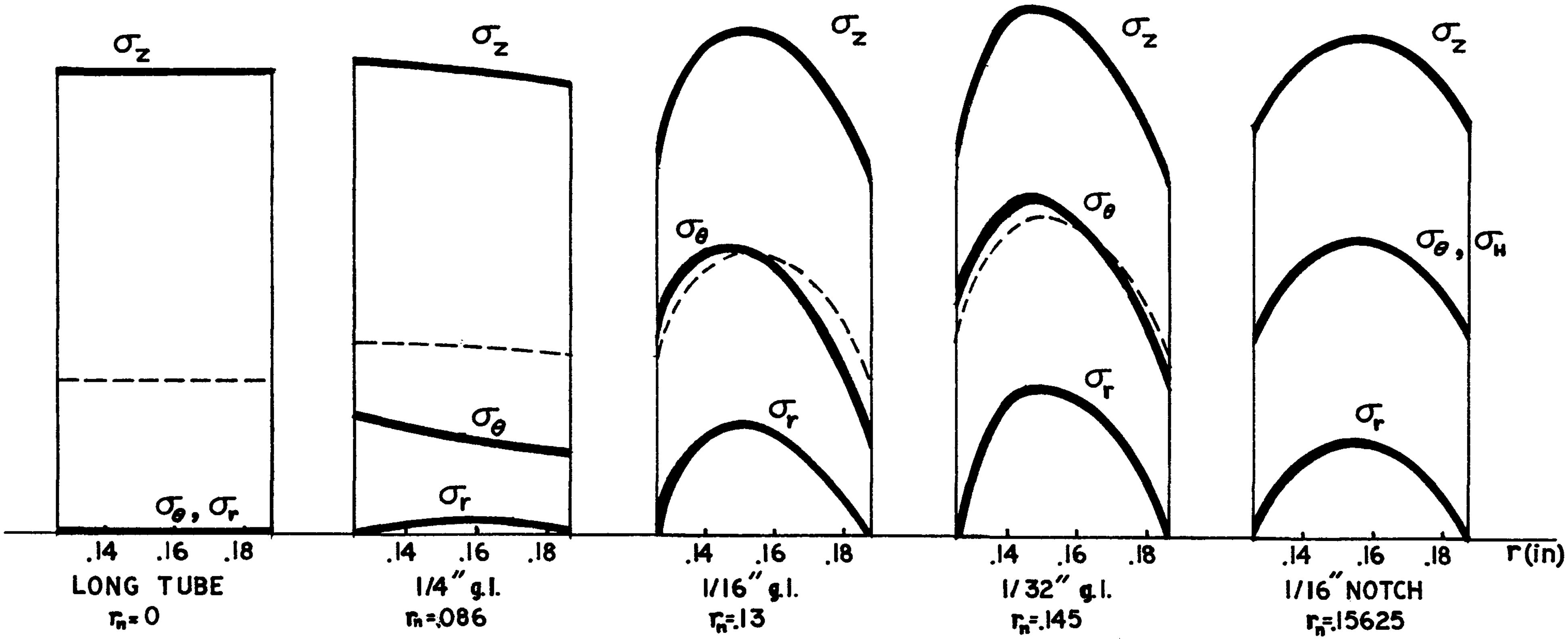


FIG 7. RADIAL DISTRIBUTION OF STRESSES $\sigma_z, \sigma_r, \sigma_\theta$.
HYDROSTATIC STRESS ------

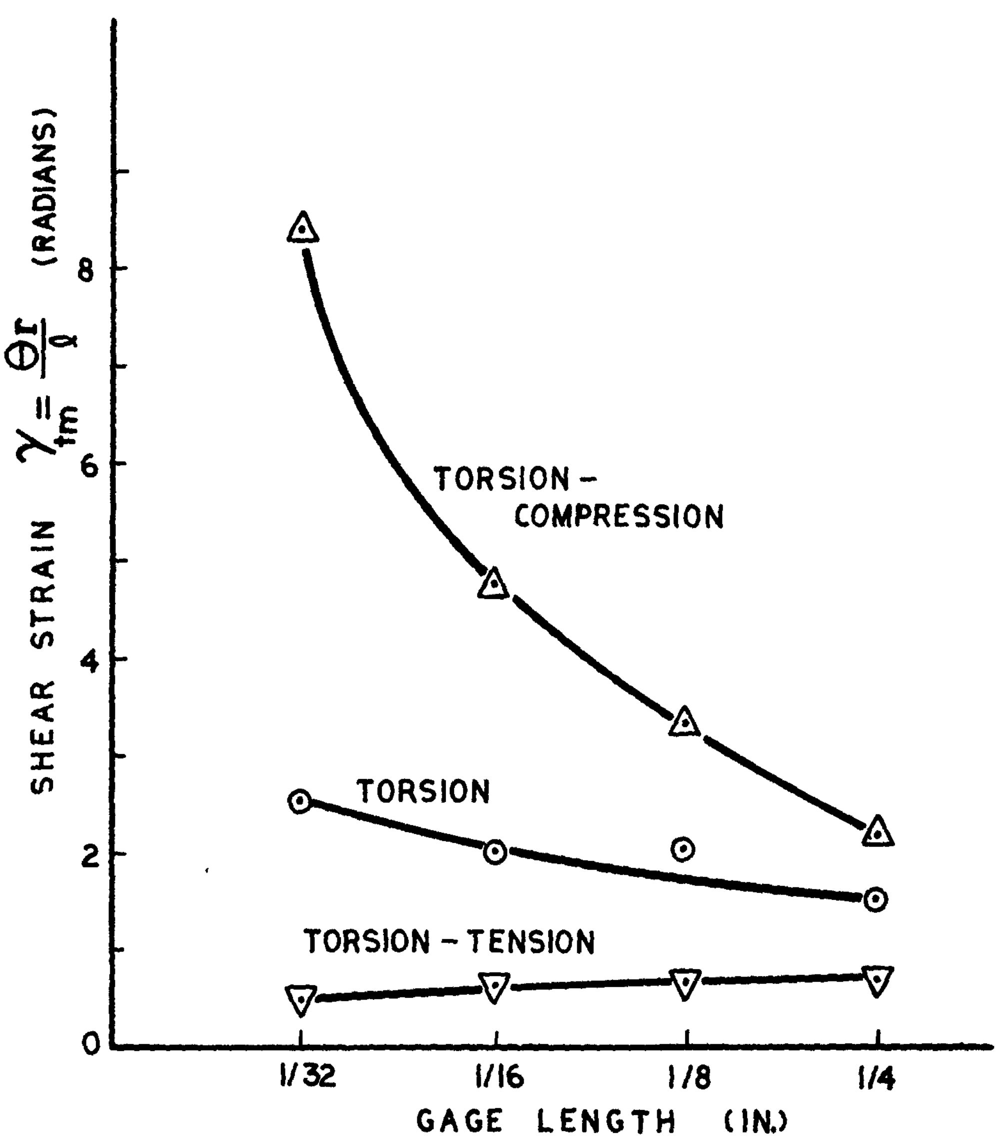


FIG 8. STRAIN AT MAXIMUM TORQUE (ONSET OF FRACTURE) IN ROOM TEMPERATURE TESTS ON INCONEL.

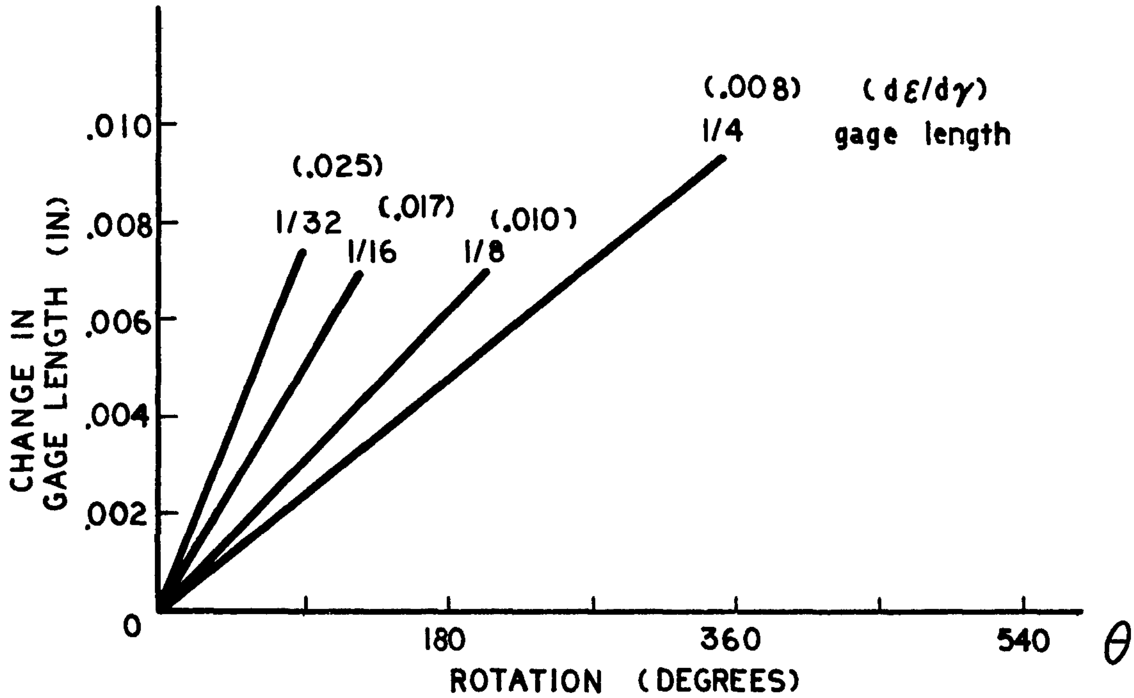


FIG 9. CHANGE IN GAUGE LENGTH WITH ROTATION IN ROOM TEMPERATURE TESTS UNDER PURE TORSION

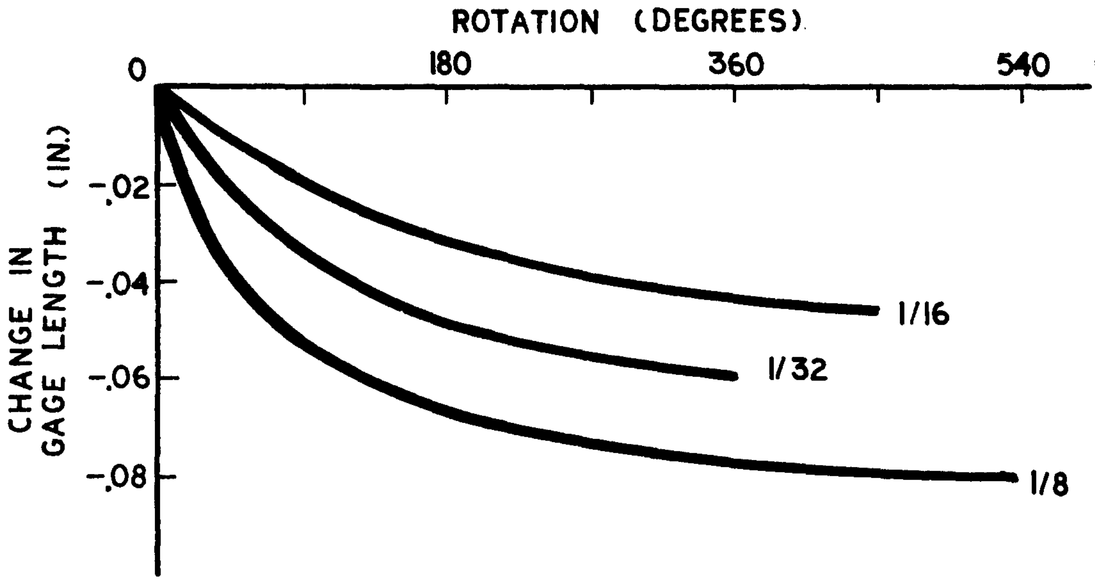


FIG 10. CHANGE IN GAUGE LENGTH WITH ROTATION IN ELEVATED TEMPERATURE TESTS (1800° F.) UNDER PURE TORSION

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4. DESCRIPTIVE NOTES *(Type of report and inclusive dates)*
Summary Report

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Kuhn, H. A.

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13. ABSTRACT

An analysis of the stresses in the axial stress-torsion test is presented by considering the gage section as a cylinder undergoing uniform axial deformation with superimposed torsion. Calculation of the stresses requires measurement of the radial deformation and penetration of the plastic region into the notch shoulders.

The hydrostatic stress exhibits a peak within the wall of the gage section which increases with decreasing gage length. The fracture strain in room temperature tests increases with decreasing gage length when a compressive axial load is applied and decreases slightly with decreasing gage length when a tensile axial load is applied. Severe distortion and large changes in length of the gage section during hot torsion testing prevent application of the analysis to high temperature tests.

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